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CAPILLARY PUMP LOOP (CPL) HEAT PIPE DEVELOPMENT STATUS REPORT

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SECTION 1. INTRODUCTION

A significant advance in heat pipe technology has recently been realized by the re-introduction of the Capillary Pump Loop (CPL) concept as a potential candidate for the management of large heat loads. The CPL heat pipe, first developed by NASA/Lewis in the mid 1960's (Reference 1), is a two-phase heat transfer device capable of transferring heat efficiently, with little temperature differential and no external power (pump) requirements. It offers large heat load carrying capacities together with substantial wicking height in gravity. It is currently being evaluated for potential application in the thermal management of large space structures (Reference 2)

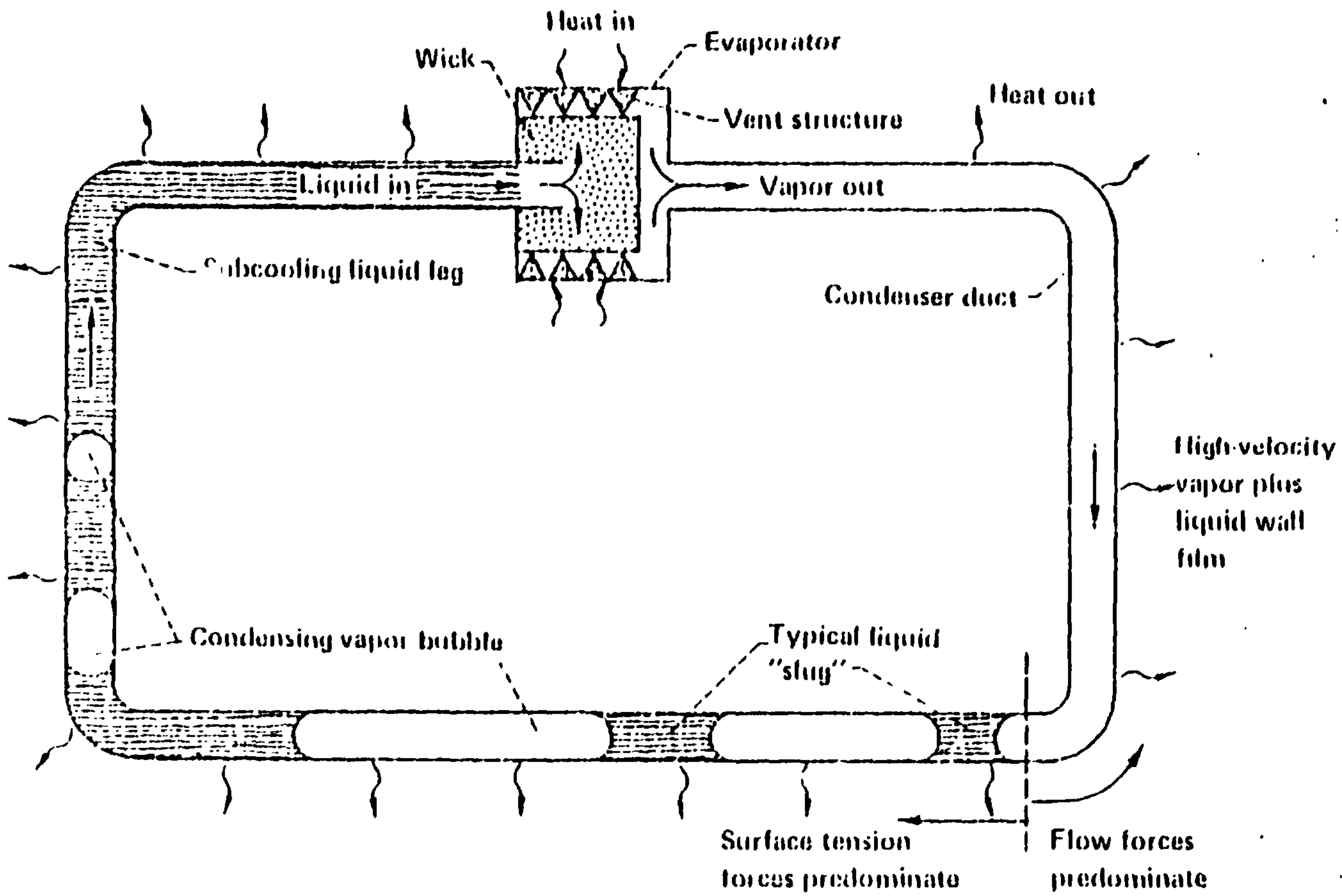
Experimental loops tested by NASA/Lewis in the mid 1960's demonstrated the feasibility, the potential high heat load capacity and low gravity sensitivity of the CPL heat pipe concept. Recently, a CPL heat pipe prototype was developed and tested for NASA/GSFC to demonstrate the practicality of such a design including the feasibility of multiple parallel evaporator/condenser zones within a single loop, heat load sharing between evaporators, liquid inventory/temperature control feature, priming under load and entrapment of non-condensible gases. The design also incorporates very long vapor and liquid return headers (approximately 32 feet) designed to demonstrate the ability of transferring heat loads over a large distance.

The primary objectives of development and test efforts conducted to date were to establish feasibility of the CPL heat pipe design. The scope of these efforts were limited and the CPL heat pipe prototype was designed and fabricated using commercially available hardware and materials. The scope of the test efforts were also limited to proof-of-concept testing. Despite these limitations, significant progress has been made in the development of the CPL heat pipe. The current development status of the CPL heat pipe is presented in the section that follows.

SECTION 2. CPL HEAT PIPE OPERATING PRINCIPLE

The basic design of the CPL heat pipe, illustrated schematically in Figure 2-1, consists of an evaporator, which includes a wick structure, and a continuous loop which is devoid of any wick material. The loop, which can be made of smooth wall tubing, provides a flow passage for the vapor, heat transfer area in the condenser and liquid return to the evaporator.

The CPL heat pipe is a continuous loop in which both the vapor and liquid always flow in the same direction. As heat is applied to the evaporator, liquid evaporates from the saturated wick and flows through the loop to the condenser zone where heat is removed. Flow in the condenser zone initially consists of high-velocity vapor plus liquid wall film which subsequently turns into liquid "slug flow". The liquid is returned to the evaporator via a subcooled liquid return header which collapses any remaining vapor bubbles. The uniqueness of the CPL heat pipe is that a wick structure is required in the evaporator zone only. Capillary action is limited to that zone, where it provides the necessary pressure differential to initiate vapor flow. In the remainder of the loop, pressure exerted by the flow of vapor on the liquid column ahead of it drives the liquid back to the evaporator by a positive "piston" action.



Source: Stenger, F. J., *Experimental Feasibility Study of Water-Filled Capillary-Pumped Heat-Transfer Loops*, NASA TM X-1310, November 1966.

Figure 2-1. Capillary Pump Loop Schematic

SECTION 3. PROTOTYPE DESIGN

The design of the CPL heat pipe, illustrated in Figure 3-1, differs significantly from the early NASA/Lewis capillary pumped loop. The major differences include:

a. Parallel Evaporators and Condensers. Parallel circuits are essential to minimize pressure drops; to accommodate multiple heat sources and heat sinks; and to facilitate the design and construction of cold plates. Pressure drop considerations are especially critical because of the limited pumping head that can be developed by capillary action.

b. Reservoir. A reservoir has been included as an integral part of the CPL heat pipe design. Its primary function is liquid inventory control. Additional features derived from the use of the reservoir include temperature control; pressure priming under load and/or against gravity; and reduced sensitivity to fluid leaks.

c. Non-Condensible Gas (NCG) Trap. A well established weakness of high composite capillary systems (such as the CPL heat pipe), is their sensitivity to the presence of non-condensible gases. The one-way action of the capillary pumped loop and the physical separation of the vapor and liquid flow channels allows the introduction of a trap to prevent the displacement of non-condensible gases into the evaporator zones.

Other features incorporated into the CPL heat pipe design include separate sizing of the vapor header and liquid return channels to optimize pressure drops; and the use of axially grooved tubing in the condenser to enhance heat transfer.

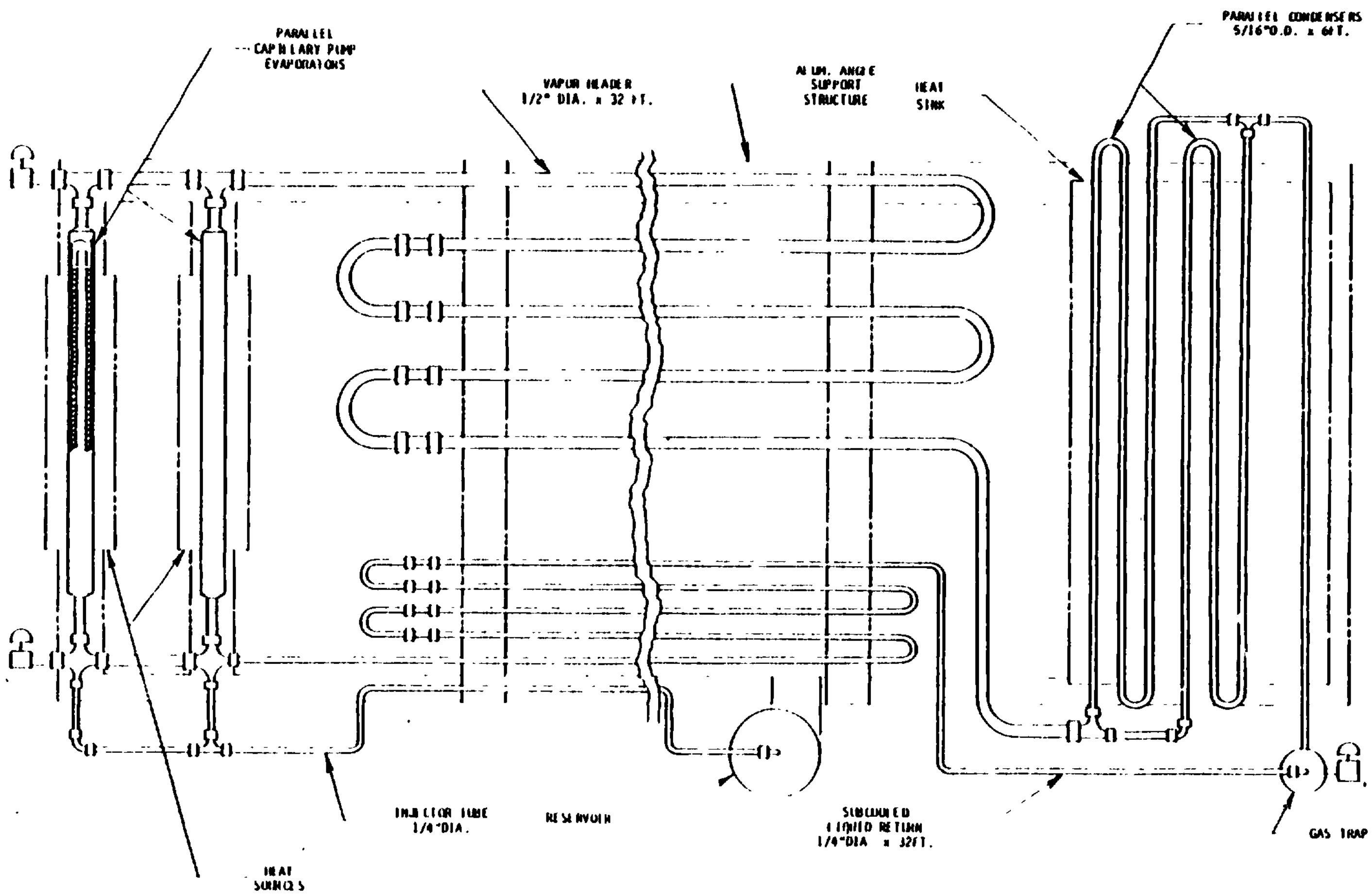


Figure 3-1. CPL Heat Pipe Prototype

3.1 DESIGN SUMMARY

A CPL heat pipe prototype (Figure 3-1) consisting of two parallel capillary pump evaporators and two parallel axially grooved condensers was developed for proof-of-concept testing. The evaporator zone and condenser zone are interconnected by a 1/2 in. diameter X 0.035 in. wall vapor header and a 1/4 in. diameter X 0.035 in. wall liquid return. Both the vapor header and the liquid return are made of smooth walled aluminum tubing bent into several passes to provide an adiabatic transport length of 9.8 meters (32 feet). The CPL heat pipe prototype design also includes a 100 cc capacity non-condensible gas trap located at the outlet of the condenser zone and a 1000 cc capacity liquid reservoir (accumulator) connected to the evaporators as shown in Figure 3-1. All tubing connections were made with stainless steel compression fittings.

Design characteristics of the CPL heat pipe prototype are summarized in Table 3-1. Theoretical performance, based on heat pipe theory (Reference 3), is shown in Table 3-2. Wick properties were derived from open air static wicking height and flow measurement tests performed with alcohol on the as-fabricated capillary pump evaporators prior to loop assembly. The analysis performed considered both laminar and turbulent vapor flow. Liquid flow analysis in the evaporator considered radial flow conditions from a smaller wick inner diameter to a large wick outter diameter, as well as the effect of the perforated liquid return tube.

The CPL heat pipe was assembled on an 8 ft. X 2 ft. test frame made of aluminum angles. Attachments were made via phenolic and nylon isolators. Aluminum heater blocks equipped with cartridge heaters and water flow channels were clamped on opposite sides of each evaporators. Water flow channels were provided for heat load sharing experiment. Heat sinking in the condenser region consisted of a grooved brass plate to which the condenser tubing was clamped down with thermal joint compound between the surfaces. The brass plate is equipped with a copper cooling coil which is connected to GSFC's Thermophysics Laboratory Tinney cooling loop.

Table 3-1. CPL Heat Pipe Prototype Design Summary

Evaporators	
• Number of Evaporators in Parallel	2
• Total Length	0.38 m (15 in.)
• Active Length	0.31 m (12 in.)
Condensers	
• Number of Condensers in Parallel	2
• Maximum Active Length Per Condenser	1.8 m (72 in.)
• Subcooled Section Length	0.61 m (27 in.)
Wick	
• Material	Alumina/Silica Felt
• Mean Fiber Diameter	2.5 Micron
• Packing Density	40%
• Effective Pumping Radius (Measured)	9.7 Micron
• Permeability (Measured)	2.6×10^{-13} m^2
• Geometry	See Figure 3-2
Evaporator Tube	
• Material	Alum. Axially Grooved Tubing
• Permeability	4.57×10^{-8} m^2
• Flow Area	6.9×10^{-5} m^2
• Geometry	Tag 54-5 (See Figure 3-2)
Vapor Header	
• Material	Alum. Smooth Walled Tubing
• Inner Diameter	1.1×10^{-2} m (0.43 in.)
• Length	9.8 m (386 in.)

Table 3-1. CPL Heat Pipe Prototype Design Summary (cont.)

Liquid Return

• Material	Alum. Smooth Walled Tubing
• Inner Diameter	4.6×10^{-3} m (.18 in.)
• Length	9.8 m (386 in.)

Condenser

• Material	Alum. Axially Grooved Tubing
• Vapor Diameter	4.7×10^{-3} m (0.185 in.)

Table 3-2. CPL Heat Pipe Prototype Theoretical Performance

Working Fluid	Freon-11
Operating Temperature	20°C
Active Length Per Condenser*	0.5 m (20 in.)
Capillary Pumping Head	3940 n/m^2
Maximum Heat Transport (Zero Elevation)	400 Watts
Maximum Static Wicking Height	0.27 m (10.5 in.)
Pressure Drop Distribution	
• Wick	1630 n/m^2
• Vapor - Evaporator	100 n/m^2
• Vapor - Header	1130 n/m^2
• Vapor - Condenser	350 n/m^2
• Liquid - Condenser	60 n/m^2
• Liquid - Return Line	560 n/m^2
• Liquid - Evaporator	110 n/m^2

* Based on observed temperature profile with the cold plate temperature at approximately - 3°C

3.2 EVAPORATOR DESIGN

Construction of the capillary pump evaporators, illustrated in Figure 3-2, consisted of 1.15 in. Tag 54-5 axially grooved aluminum tubing packed with aluminum/silica fiber wick. The wick was made by cutting washers from 1 in. thick felt that was inserted and packed into the Tag 54-5 tubing to an approximate density of 40 percent. Approximately 140 washers were inserted into each of the capillary pump evaporators. Located in the center of the wick is a 3/8 in. diameter perforated liquid return with a 1/4 in. diameter reservoir injector tube located inside the 3/8 in. liquid return. The compacted wick material is retained by an aluminum retainer fitting at the vapor outlet end and a press fitted aluminum washer at the liquid return end. Welded aluminum end fittings complete the assembly of the capillary pump evaporator.

3.3 CONDENSER DESIGN

Axially grooved tubing was used in the design of the condensers to enhance heat transfer characteristics. The condenser zone consists of two 1.8 meters (6 ft.) parallel paths and a 0.6 meters (2 ft.) subcooled zone.

3.4 LIQUID INVENTORY MANAGEMENT

Stable operation of the CPL heat pipe requires complete condensation of the vapor phase before the working fluid enters the evaporator zone. In addition, liquid entering the evaporator must be at a temperature lower than saturation. This is because a finite amount of heat, conducted through the liquid return end fitting, would cause vapor bubbles to be generated in a saturated liquid. Since these vapor bubbles cannot pass through the wick structure, they would accumulated within the liquid return line until the resulting blockage would deprime the wick. To avoid the above conditions, a portion of the condenser must be allocated as a subcooled region. This can be accomplished by charging the CPL heat pipe with sufficient liquid to ensure a partially blocked condenser at all

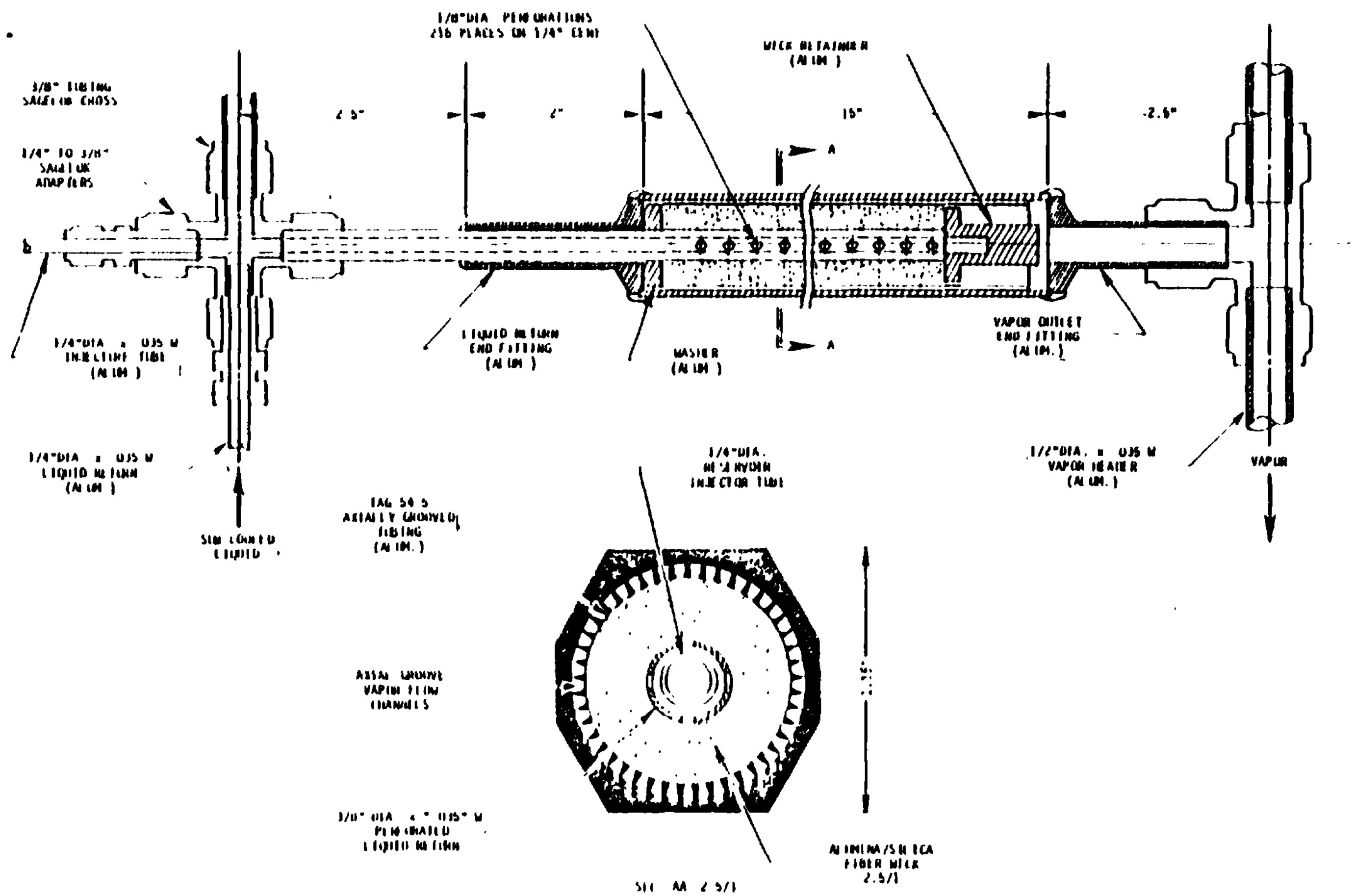


Figure 3-2. Capillary Pump Evaporator

times. Predetermined charging of the CPL heat pipe, however, presents several problems which include the following:

a. Precise fluid inventory calculations and accurate charging procedures that would be required since condenser zone volume is small compared to total volume. That is, even small variances in total inventory will significantly affect extent of condenser blockage.

b. Liquid entrapment in the vapor header can significantly affect condenser zone blockage. The amount of entrapment, if any, cannot be predetermined.

c. If a non-condensible gas (NCG) trap is used in the system design, liquid inventory will be displaced as NCG's are collected. Additional blockage of the condenser by the displaced liquid will cause a performance degradation.

d. Fluid densities vary as a function of temperature. A CPL heat pipe charged to have a partially blocked condenser at low operating temperatures could be completely blocked at high operating temperatures. The operating temperature of a CPL heat pipe with a fixed charge would be limited depending on the relative volume of the condenser zone as compared to the remainder of the system.

To avoid the above problems, a reservoir (accumulator) has been incorporated as part of the CPL heat pipe design. As illustrated schematically in Figure 3-3, distribution of the liquid between the reservoir and the loop is maintained by a pressure balance between the saturation pressure in the reservoir and the saturation pressure in the loop. Any change in loop operating temperature due to variations in condenser blockage caused by the above factors or variations in heat load and/or sink temperature will cause a pressure imbalance which will result in a displacement of liquid into or out of the reservoir. Condenser blockage is thus increased or decreased until equilibrium is restored.

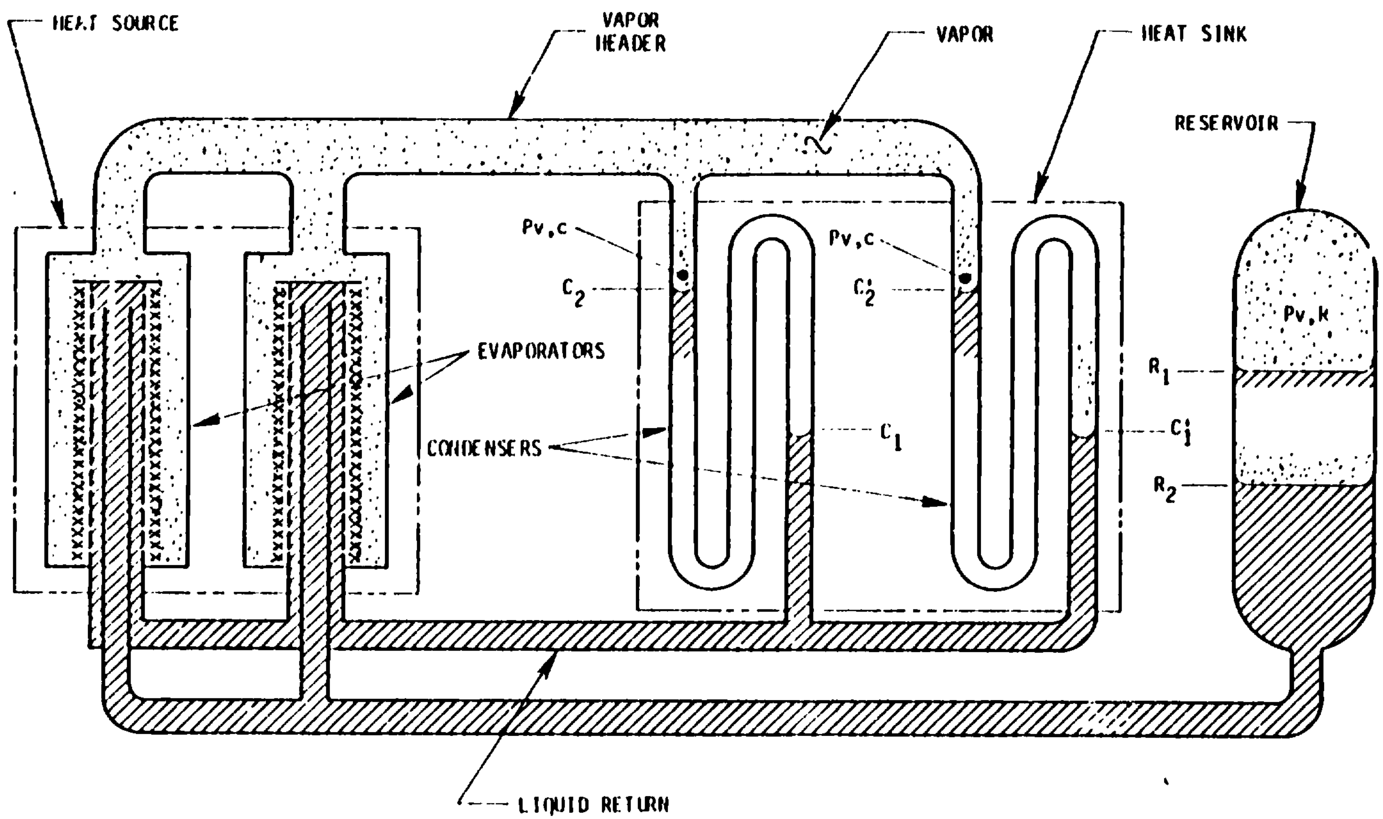


Figure 3-3. Liquid Inventory Distribution

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In effect, the reservoir saturation conditions will control the loop saturation conditions as long as:

a. The reservoir is of sufficient volume to accommodate the range of liquid displacement required to maintain variations in operating conditions.

b. The reservoir is only partially filled under all conditions.

c. A partially blocked condenser is maintained at all times.

To satisfy the last condition, the reservoir must be maintained at a temperature which is equal to or greater than the temperature at which liquid displacement from the subcooled region will occur. Any lower reservoir temperature will drain an excessive amount of liquid from the loop to a point where subcooling conditions are no longer satisfied and the CPL heat pipe will deprime. Although the reservoir can be operated at any range of temperature, a single set point designed to accommodate maximum heat load/maximum sink condition will satisfy all other operating conditions. Thus, a simple controller will satisfy CPL heat pipe reservoir requirements.

3.5 TEMPERATURE CONTROL

The need to maintain temperature control of the reservoir is predicated by the need to control liquid inventory in the CPL heat pipe for reasons previously discussed. Although a temperature controlled reservoir adds complexity to the CPL heat pipe design, the benefits derived far outweight the disadvantages. In addition to the ability to compensate for any number of variables in the control of liquid inventory, a temperature controlled reservoir provides the following additional benefits:

a. Operating temperature of the CPL heat pipe is controlled by the set point of the reservoir. Temperatures of components mounted on the CPL heat pipe are automatically maintained over the entire range of operating conditions.

b. Complete shut-down of the CPL heat pipe can be achieved. Component temperature will be maintained as long as sufficient power is available to compensate for parasitic heat leaks.

c. Automatic shut-down of the condenser under conditions where heat load sharing between components is required.

The temperature control feature is derived from a pressure balance that must be maintained between the saturation pressure in the reservoir and the saturation pressure in the loop. As stated earlier, any change in loop temperature due to variations in heat load and/or sink temperature will cause a pressure imbalance which will result in a displacement of liquid into, or out of the reservoir. Condenser blockage is thus increased or decreased until equilibrium is restored. Control of the reservoir temperature automatically controls the vapor temperature i the CPL heat pipe. The CPL heat pipe thus operates like a variable conductance heat pipe with the exception that liquid blockage, instead of non-condensible gas blockage, is used to vary the active length of the condenser.

The fact that saturation pressure balance must be maintained also provides the ability for complete shut-down of the loop and heat load sharing between components. If power from components is not sufficient to maintain saturation conditions, complete flooding of the loop or total depletion of the reservoir inventory (depending on relative volumes) will result. In either case, shut-down of the CPL heat pipe is achieved. Similarly, if heat sharing is required between components, partial or complete shut-down of the condenser will occur in order to maintain saturation pressure balance. The limiting factor in load sharing is the need to maintain subcooled inlet conditions into the active evaporators.

3.6 PRESSURE PRIMING

A heated reservoir provides the ability to prime the CPL heat pipe under load and/or against gravity. Priming can be achieved by raising the reservoir temperature to develop sufficient pressure to allow displacement of liquid into the evaporators.

In a typical dry-out of the CPL heat pipe, either thermal or mechanical, depletion of the liquid inventory in the evaporators causes a total blockage of the condenser zone which extends into the vapor header. By maintaining the vapor header temperature at higher than evaporator dry-out temperature, and by cycling the reservoir to a temperature above the dry-out temperature, pressure priming of the evaporators can be achieved. Two factors make this priming mechanism possible: 1) the high saturation pressure established in a heated vapor header which allows liquid to be injected into the evaporators as if the evaporators were subcooled and 2) complete shut-down of the loop by virtue of the blocked condenser zone which allows priming under an artificial "no-load" condition. Priming under load is thus possible as long as the rate of evaporator temperature rise under the dry-out condition is sufficiently slow to allow the temperature cycling of the reservoir before the dry-out temperature exceeds vapor header temperature and/or before physical damage of the loop and the components mounted to the loop. Also, heat must be applied to the vapor header to maintain an elevated temperature in that zone. The power level required can be minimal as long as the vapor header is well insulated. During normal operation, power applied to the vapor header helps prevent liquid entrapment. The need to maintain power on the vapor header can be eliminated for pressure priming purposes with a reservoir fluid inventory capable of flooding the entire loop.

3.7 NON-CONDENSIBLE GAS TRAP

Any of the non-condensible gases in the CPL heat pipe will be swept along in the flow stream of the working fluid. Since they cannot be condensed, they begin forming small bubbles in the condenser. Eventually the bubbles will migrate into the evaporators where surface tension forces at the wick interface will prevent further migration. If sufficient quantities of non-condensible gases are present in the system, accumulation of bubbles in the evaporators will continue until the CPL heat pipe deprimes.

To ensure reliable CPL heat pipe operation, non-condensible gases must be reduced to sufficiently low quantities or they must be trapped in an area where they will not detrimentally affect performance. Past experience with conventional heat pipes has demonstrated the impracticality of producing gas free heat pipes. The problem is compounded in the CPL heat pipe due to extremely long tubing lengths and tightly packed wick structure which makes it impracticle to properly evaluate the system prior to charging. In addition, reflux boiler methods of gas separation used in conventional heat pipe processing cannot easily be applied to the CPL heat pipe due to its complex geometry and because gases tend to collect in the evaporator (instead of the condenser) where they cannot easily be removed.

For these reasons a non-condensible gas trap was introduced into the design of the CPL heat pipe. The gas trap is located at the exit of the condenser and is designed (Figure 3-4) to separate gas bubbles from the liquid by means of buoyancy forces exerted on the gas bubbles. This design was selected for its simplicity and ease of implementation.

In zero-g operation, a mechanism other than buoyancy will be required to separate non-condensible gases from the liquid. One approach will be to use a gas trap design which incorporates a wick structure designed to prevent further migration of gas bubbles by surface tension forces at the wick interface.

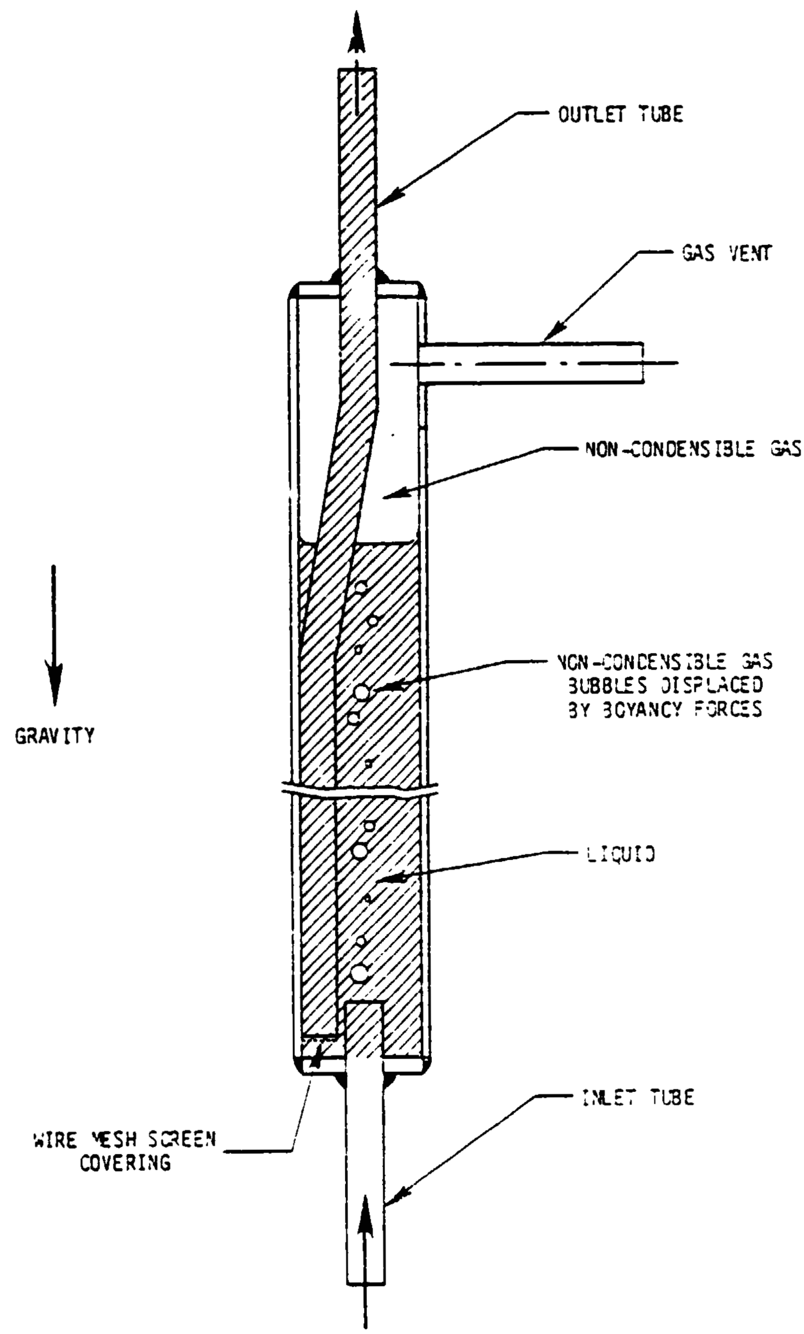

Figure 3-4. Non-Condensible Gas Trap.

SECTION 4. PROOF-OF-CONCEPT TEST

CPL heat pipe proof-of-concept tests have been conducted on several occasions during the past 10 months. Testing was limited in scope and design primarily to establish feasibility of design, to obtain an understanding of operating behavior and to establish basic performance characteristics. Results of tests conducted on three separate occasions are summarized below.

4.1 TEST SET-UP

Prior to testing, the CPL heat pipe was briefly evacuated with a mechanical pump and charged with commercially available Freon-11. No special attempt was made to minimize non-condensible gases.

The CPL test set-up (Figure 5-1) included individual control of evaporator power to establish the ability of operating parallel circuits at different power levels. Provisions were also made to cool the evaporators to establish the feasibility of heat load sharing. The set-up also included a vapor header heater designed to prevent liquid entrapment and to provide a back pressure during priming, and a reservoir heater for temperature control and pressure priming cycles. At the condenser end, the parallel condenser tubes were clamped to a cold plate connected to the GSFC Thermophysics Laboratory Tinney coolant loop. The entire loop was insulated with fiberglass insulation blankets with the exception of the cold plate which was left un-insulated to obtain visual observation of the liquid/vapor interface location in the condenser. Visual observations were made possible by maintaining the cold plate at a low enough temperature to form frost which was subsequently thawed as the vapor front advanced into the condenser zone. Finally, instrumentation of the CPL heat pipe prototype consisted of copper-constantan thermocouples (see Figure 5-1 for locations).

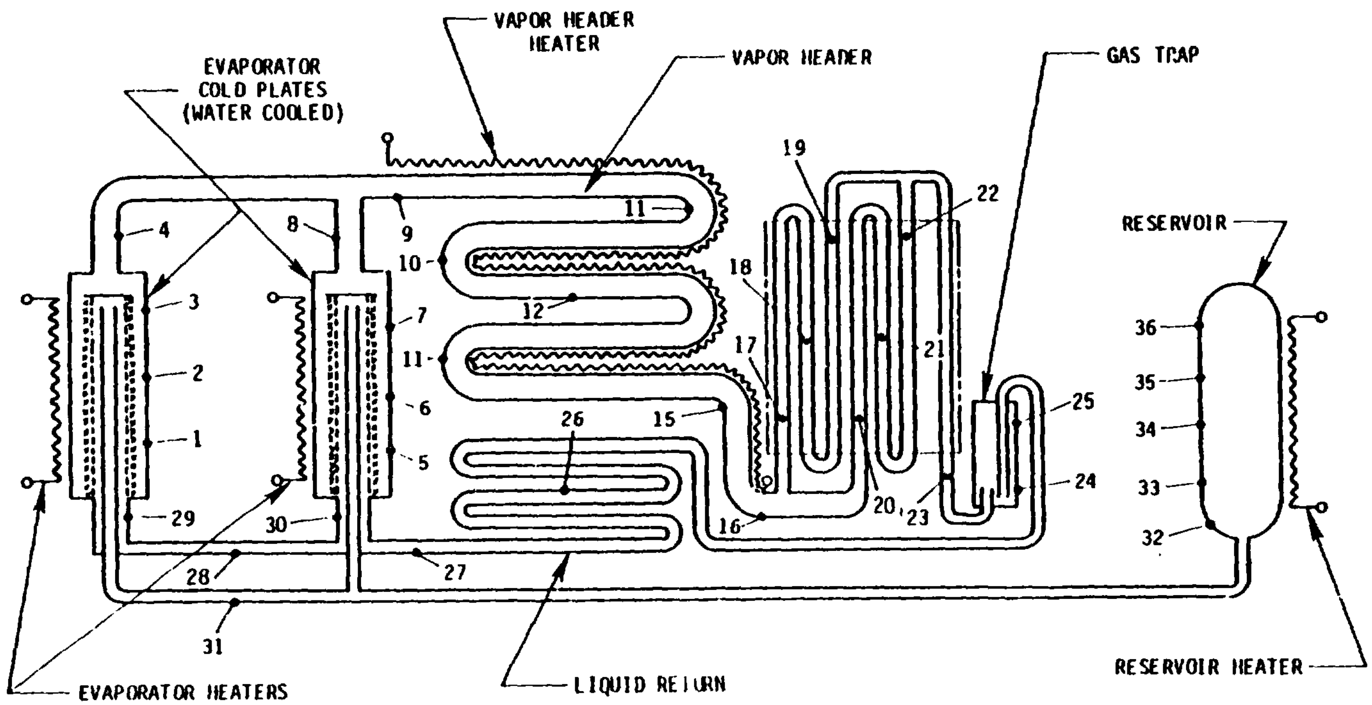

Figure 4-1. CPL Heat Pipe Test Set-Up

4.2 HEAT TRANSPORT PERFORMANCE

The CPL heat pipe was first operated at the end of August 1981. After an initial success, start-up of the CPL proved to be difficult. An excessive amount of non-condensible gases was determined to be present in the system. After several bleeding operations, the CPL heat pipe was successfully operated at power levels in excess of 200 watts and at elevations of 2.0 inches. Performance tests were repeated on several occasions substantiating the CPL heat pipe's capability of transferring large heat loads over long distances and at substantial elevations. The initial tests also indicated that a parallel circuit design is feasible and that unbalanced heat loads can be applied to the evaporators. In addition, the fact that the CPL heat pipe was successfully operated with minimal attempts to control non-condensible gases indicates that the problem of gas bubbles may be circumvented with this design.

Performance tests were repeated at the end of September/beginning of October 1981 and in February 1982. Results, including evaporator and condenser conductance, are summarized in Table 4-1. The September/October test results essentially duplicated initial test results. Capacities in the range of 200 watts at an elevation of 2.0 inches were obtaned. On the basis of the data obtained, extrapolated zero elevation performance of 250 watts was achieved as compared to theoretical predictions of 400 watts. An attempt, at the end of this test effort, to obtain performance data at 4.0 inches of elevation was unsuccessful.

During the September/October performance testing, it became apparent that significant quantities of non-condensible gases were still present in the system. This determination was made on the basis of the relatively high loop operating temperature as compared to the reservoir set point temperature. After several reservoir bleeding operations, a significant reduction in loop operating temperature with respect to reservoir temperature was noted. Further attempts to remove non-condensible gases from the system were discontinued because of the inability to clearly establish a gas interface in the reservoir. This was due to the fact that the

Table 4-1. CPL Heat Pipe Prototype Performance Summary

	Units	FREON-11 (Measured) 9/18 - 10/81 Test Data	FREON-11 (Measured) 2/82 Test Data	Ammonia (Extrapolated)
Heat Transport				
• Maximum Heat Load Capacity Achieved	Watts	200 Watts @ 2 in.	350 Watts @ 1.5 in.	--
• Estimated Maximum Heat Load Capacity at Zero Elevation	Watts	250 Watts	400 Watts	3400 Watts
• Heat Transport Capacity	Watts	2500 W-M	4000 W-M	34,000 W-M
Heat Transfer				
• Evaporator Film Coefficient	$BTU/HR\text{-}FT^2\text{-}^oF$	250-500	250-500	1400-2800
• Condenser Film Coefficient	$BTU/HR\text{-}FT^2\text{-}^oF$	750-1500	750-1500	4000-8000

reservoir was heated along its entire length which masked any gradients developed by non-condensible gas/vapor separation.

After a four-month inactive period, the CPL heat pipe was reactivated in February 1982. The primary objective was to determine if any residual non-condensible gases were present in the system, to remove the gases if possible and to determine any effect gas reduction would have on performance. A new heater, confined to the bottom of the reservoir, was installed and a positive indication of non-condensible gases was obtained. After repeated reservoir bleeding operation, it became apparent that the liquid inventory was being depleted and the CPL heat pipe was charged with additional Freon-11. After the reservoir bleeding operation was completed, previously achieved performance could not be duplicated. The CPL heat pipe was then bled at the non-condensible gas trap valve, to ensure that it was not totally blocked, and at the valve located at the end of the liquid return line (see Figure 3-1 for valve location). A positive displacement of non-condensible gas was obtained at both locations. The CPL heat pipe was then performance tested at an elevation of 1.5 inches and a heat transport capacity of 340-350 watts was achieved. The performance level was repeated during several test runs. The measured performance extrapolates to approximately 400 watts at zero-elevation which is identical to theoretically predicted performance. The small elevation difference, compared to previous testing, cannot explain the performance difference. Possible explanations include bleeding of non-condensible gases, additional liquid charge (previous charge inadequate due to repeated reservoir bleeding operations) and/or Freon-11 is better able to bridge (form a slug rather than a puddle) the inner diameter of the liquid return at lower elevation.

The most likely explanation is that removal of non-condensable gases to a low enough level allowed the CPL heat pipe to be operated without premature depriming. Displacement of non-condensible from the liquid return may be especially troublesome because of the very long liquid return and the possibility of non-condensible gas bubble entrapment due to the multiple pass arrangement used in this set-up. Since alternate lengths of the liquid return are tilted downward, the liquid return must act against buoyancy forces to displace any gas bubbles. In addition, Freon-11 wicking height properties are very low, making it difficult for the fluid to gap the inner diameter of the liquid return by capillary action. It is thus possible for the liquid return to have a tendency to by-pass non-condensible gas inclusions, making the displacement of gas bubbles, by entrapment, a difficult and slow process. During a significant number of tests where burnout occurred at relatively low power inputs, the amount of watt-hours transported prior to burnout were most often in excess of that required to displace total loop liquid inventory at least once and often twice. This would tend to support the possibility that non-condensible gases trapped in the liquid return were slowly migrating and hindered the ability to achieve full capacity in early tests; and that the displacement of such gases from the liquid return is a very slow process due to entrapment in the multiple pass set-up. Future tests will consider this possibility including means of effectively removing residual non-condensible gases from the liquid return and means of determining the effectiveness of the non-condensible gas trap.

4.3 HEAT TRANSFER PERFORMANCE

Evaporator and condenser film coefficients derived from performance tests conducted to date are summarized in Table 4-1. A range of values is indicated to account for uncertainties in temperature measurements, location of thermocouples, establishment of a reference saturation vapor temperature, effects of non-uniform heat input/output and determination of active zone in the condenser. Establishment of a reference saturation vapor temperature is made difficult since a heated vapor header is used in this set-up. Determination of the active condenser tone was also difficult since its not possible to establish the length of annular flow and slug flow zones since both are subcooled and cannot be clearly detected with thermocouples.

Accurate determination of heat transfer characteristics will require controlled test conditions including instrumentation more suitable for this purpose. Nevertheless, a high heat transfer efficiency of the CPL heat pipe is clearly indicated from the data obtained to date. Evaporator film coefficients are two to four times those of conventional axial groove heat pipe designs. This performance improvement is consistant with observations made by Saaski for inverted miniscus evaporator designs (Reference 4). Very efficient condenser performance is also apparent from the observation that only short, fully active zones were required to reject large quantities of heat. Minimum lengths associated with subcooled regions (i.e., annular, slug and all liquid phase zones) have yet to be established; however, since they represent low heat dissipation areas, they should not significantly affect an overall system design with regards to radiator surface area allocation.

4.4 PRESSURE PRIMING

An important design feature of the CPL heat pipe is the ability to establish conditions favorable for pressure priming and to initiate a priming cycle on command. The CPL heat pipe was subjected to a series of priming cycles during performance testing to determine priming characteristics and to establish conditions necessary to achieve reliable and consistant priming.

The ability of the CPL heat pipe to prime under load and/or against gravity is illustrated in Figure 4-2. After an initial steady state was established at an elevation of 2.0 inches and evaporator power input of 100 watts, power was then increased to 180 watts. At this power level, evaporator dry-out was indicated. Power to the evaporators was reduced to 20 watts, to avoid rapid and excessive temperature rise of the evaporators due to their small thermal mass, and, simultaneously, the reservoir was cycled to a temperature above that of the evaporators. The reservoir was than allowed to cool down to its initial set point. Once recovery was indicated, noted by a reduction in evaporator temperature, evaporator power was then increased to 100 watts to confirm priming. Subsequently, power was stepped up to 180 watts at which point the test was terminated. The priming cycle illustrated in Figure 4-2 was repeated on at least six different occasions during the September/October test effort, demonstrating the potential of CPL heat pipe priming mechanism. Additional pressure priming evaluation tests were conducted in February

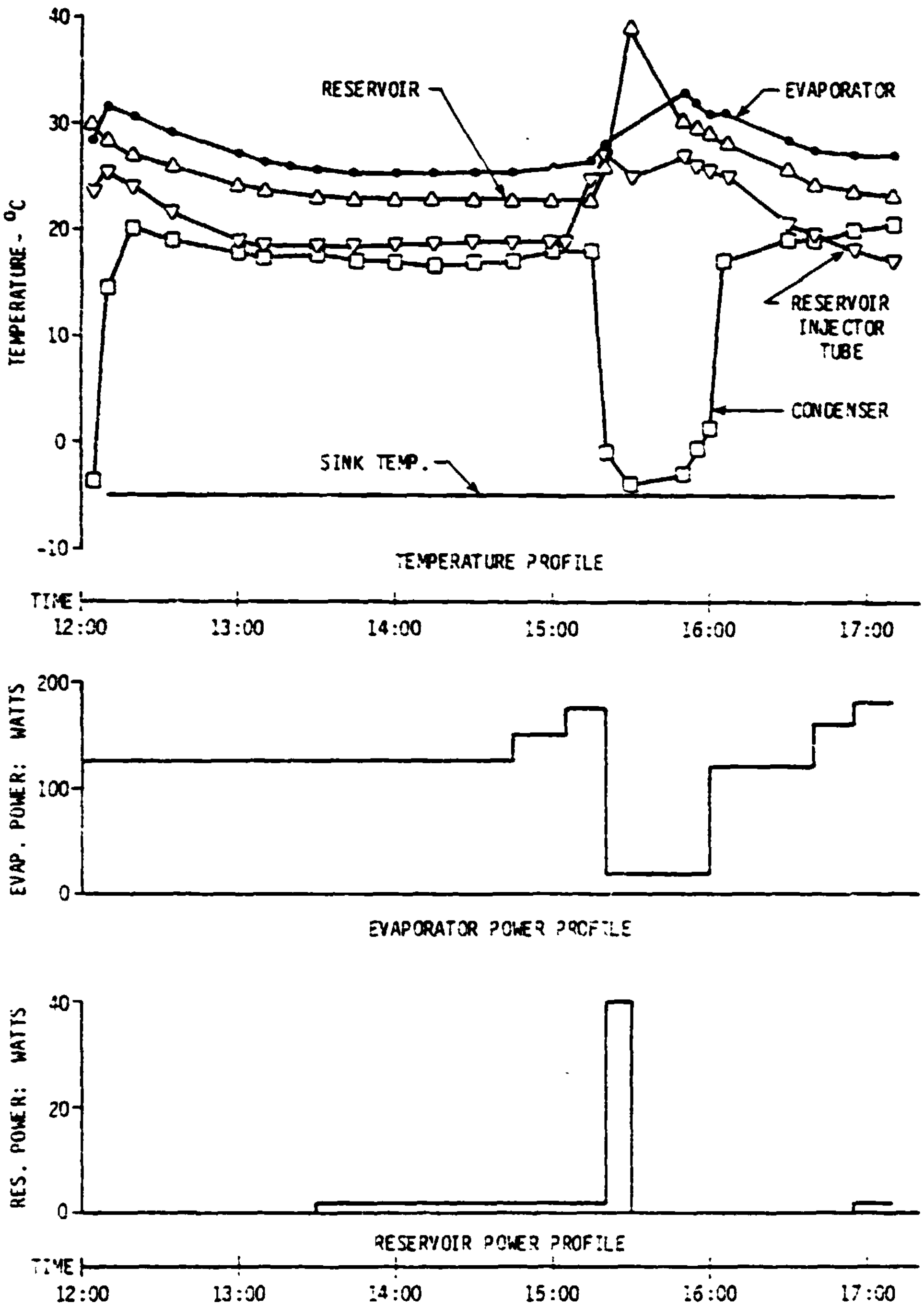

Figure 4-2. Typical Recovery Under Load

1982. Priming was judged to be successful in two out of four attempts made on the same day. Review of the data after testing indicated some differences between the successful and the unsuccessful cycles. This included the fact that the reservoir temperature was not allowed sufficient time to cycle down prior to power application to the reservoir during the unsuccessful cycles. Another observation is that a significant amount of watt-hours were transported (at least twice that required to circulate total loop inventory) prior to burnout indications after the two unsuccessful priming cycles, which may indicate the existance of some residual non-condensible gases in the liquid return line and/or that the non-condensible gas trap is ineffective.

Evaluation of pressure priming to date has been made difficult by the fact that data was obtained manually. Since a number of factors can influence priming and since time constants may be significant, additional testing and evaluation with improved and quicker data gathering methods will be required before the reliability of pressure priming can be established.

Although pressure priming has yet to be achieved on a repeatable and consistant basis, successful priming has been achieved on a significant number of occasions indicating that once the controlling factors are established and understood, a reliable procedure may be possible. Observations made to date which tend to support this priliminary conclusion include the fact that a significant rise in reservoir line temperature (T/C #31, Figure 4-1) and shut-down of the condenser have consistantly been noted prior to any noticeable rise in evaporator temperature. In addition, a rise in reservoir temperature was noted indicating that vapor and/or hot liquid are displaced into the reservoir during dry-out. The ability to detect dry-out before significant evaporator temperature rise should prove valuable in allowing priming cycles to be initiated with minimum reduction in applied system power. Condenser shut-down and displacement of vapor/liquid into the reservoir indicate that conditions favorable to pressure priming are being established during dry-out. Back-flow into the reservoir also indicates that displacement of non-condensible gas bubbles may be possible.

4.5 TEMPERATURE CONTROL

Temperature control capabilities of the CPL heat pipe is illustrated in Figure 4-3. With 120 watts applied to the evaporator, a reservoir set point temperature of 23°C and a sink temperature of -5°C, the condenser is partially opened maintaining an evaporator temperature of approximately 25°C. As the sink is cycled to -30°C, an early total shut-down of the condenser is achieved. At a sink temperature of +15°C the condenser is fully open. No noticeable change in evaporator temperature can be detected throughout the entire cycle. Similar temperature control characteristics were observed with respect to power variations. As expected, a slight rise or drop in evaporator temperature that is proportional to evaporator conductance and power input was noted during power variation cycles. These tests verified the ability using the reservoir to control the temperature of the CPL heat pipe.

4.6 HEAT LOAD SHARING

The ability to share heat between components represents a significant advantage for power savings in space systems applications. Results of a test conducted to establish feasibility of heat load sharing in the CPL heat pipe is shown in Figure 4-4. Initially, power was applied in increments to each evaporator until a total 300-watt load was achieved. This stepwise increase in power demonstrates the ability of the CPL heat pipe to control temperature as a funtion of power input. After the initial rise to operating temperature, only small temperature increments resulted with large power increases. The power increment from 100 watts per evaporator to 150 watts per evaporator resulted in only 1.5°C rise in evaporator temperature which is consistant with the measured range of evaporator film coefficients. At the 300-watt input, a fully open condenser condition (with respect to the relative location of T/C #17) is indicated. As can be seen, the condenser temperature is only 1°C below

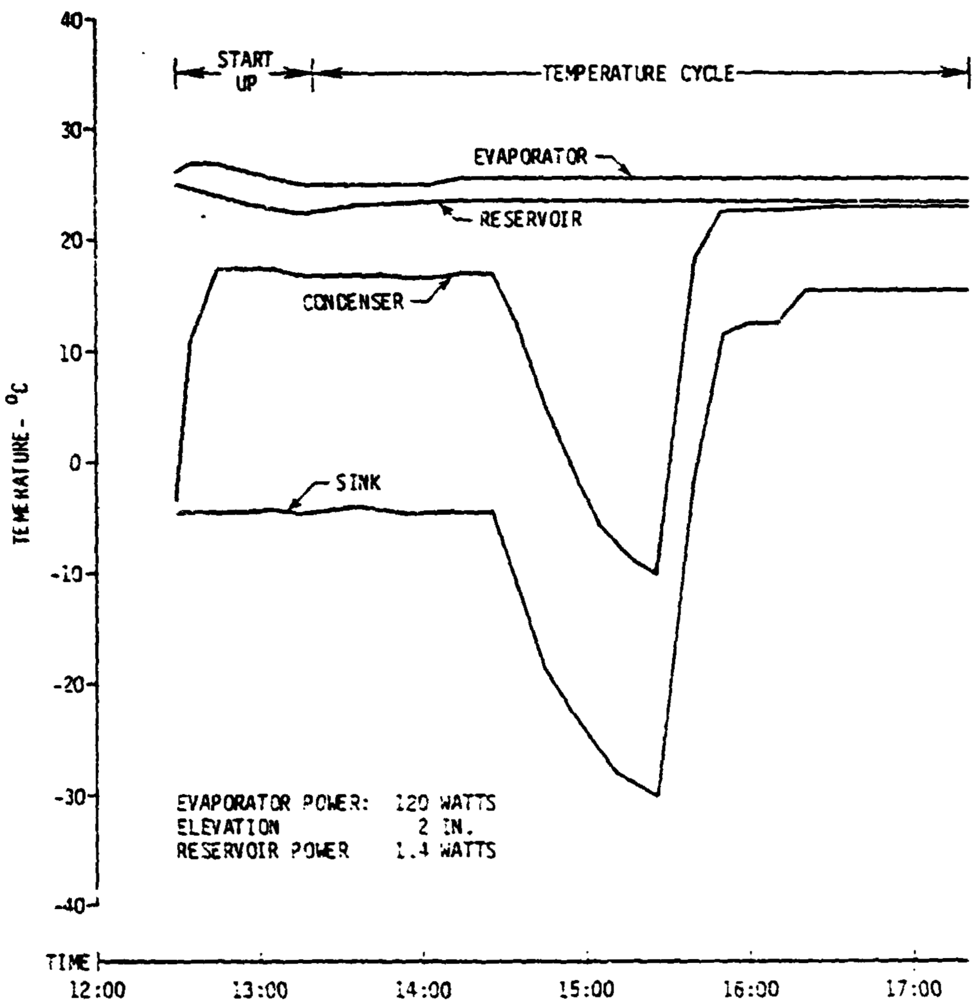

Figure 4-3. Temperature Cycle Profile

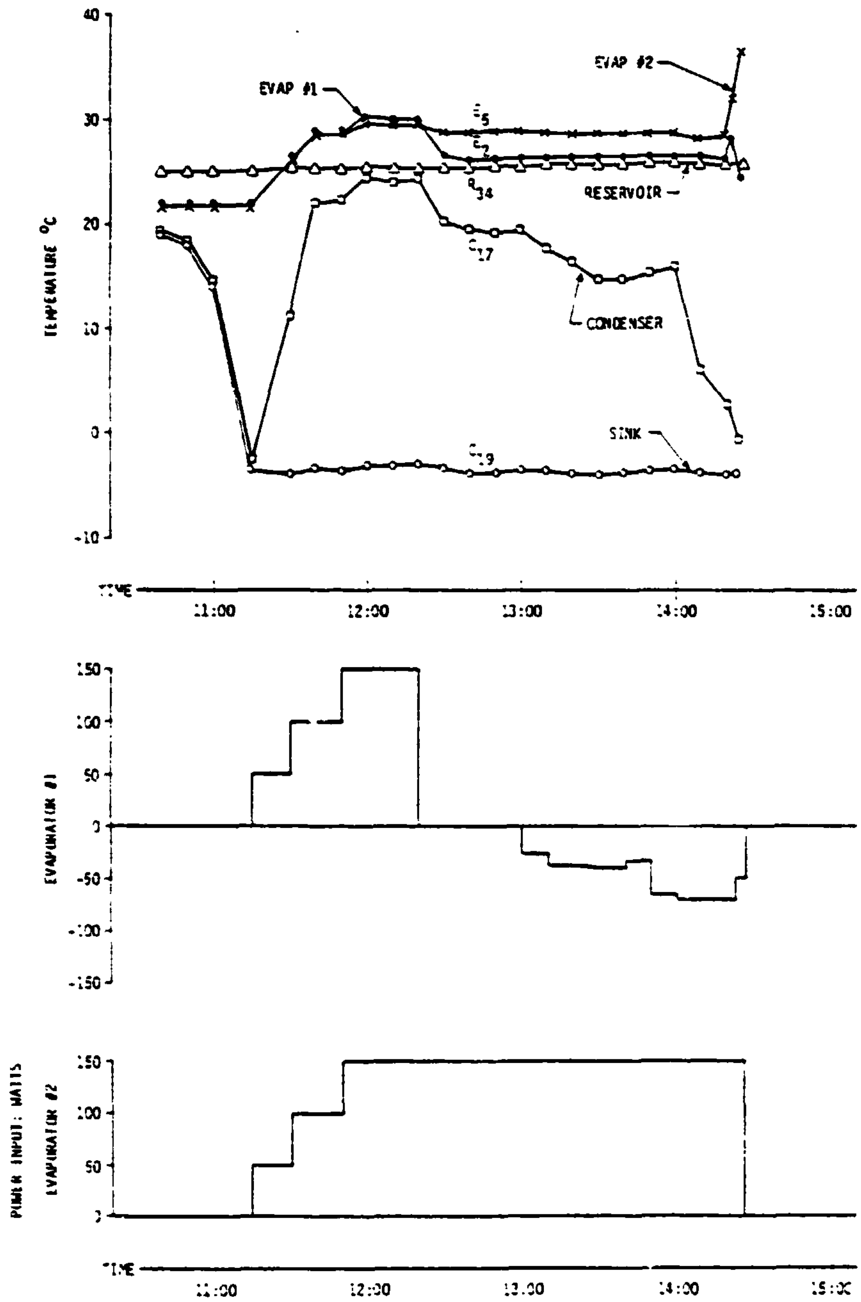

Figure 4-4. Heat Load Sharing Test Profile

that of the reservoir saturation set point temperature indicating a very high film coefficient in the fully active section of the condenser.

As the power to one of the evaporators was shut-off, its temperature reduced to a point slightly above that of the reservoir set-point. A temperature drop in the second (active) evaporator operating temperature is also apparent. Although this is yet to be fully explained, at least part of this temperature drop can be accounted for by the reduction of vapor pressure drop between the evaporator and condenser resulting from the smaller total applied heat load. A reduction in condenser temperature is also noted indicating the condenser is no longer fully active to the point at which the condener thermocouple is located.

When water cooling was initially applied (at approximately 40 watts) to the inactive evaporator, no significant change in either evaporator temperatures could be detected. The inactive evaporator continued to operate isothermally and a further reduction in condenser temperature, indicating additional shut-down, was apparent. It was also noted, as might be expected, that subcooling temperature entering the active evaporator increased as cooling of the inactive evaporator was initiated.

As cooling to the inactive evaporator was increased to 70 watts (nearly 50 percent of the applied heat to the active evaporator), a significant decrease in condenser temperature was noticed combined with a slight decrease in active evaporator temperature. The inactive evaporator remained isothermal with no apparent change in its operating temperature. After 10 minutes of operation at the increased cooling load, inlet temperature to the active evaporature increased to saturation temperatures which was followed by a depriming and burnout of the active evaporator.

The above behavior can be explained by the fact that the impedance to vapor flow between evaporators is much lower than the impedance to the condenser. Vapor, therefore, will flow preferentially between evaporators and, since pressure equilibrium with respect to the reservoir set point must be maintained, a condenser shut-down results, as cooling in the inactive evaporator is increased. In all but one respect, preferential shutdown of the condenser is an ideal condition for heat load sharing. However, the problem that develops is that, since there is little if any liquid blockage of the inactive evaporator, the liquid returned to the loop is at saturation temperature. As discussed earlier, some subcooling is required for proper operation of CPL heat pipe. Heat load sharing becomes limited by the condition of the liquid returning to the active evaporator.

Additional tests and evaluations will be required not only to confirm the above observations, but to establish more clearly the limits that might be imposed on heat load sharing by subcooling considerations. Also, evaporator designs with reduced sensitivity of subcooling and the ability to provide some means of subcooling between evaporators should be investigated before further conclusions on heat load sharing can be formulated.

SECTION 5. SUMMARY AND CONCLUSIONS

Despite the limited level of development and test efforts conducted to date, significant progress has been made in establishing the CPL heat pipe as potential candidate for systems application requiring large heat load carrying capacities over long distances. Current CPL heat pipe development status is summarized in Table 5.1. All aspects of the CPL heat pipe design have been demonstrated at least in principle. Heat load carrying capacities during proof-of-concept tests are consistant with theoretical predictions. Multi-kilowatt capacities with Freon-11 and an order of magnitude, better performance with Ammonia can be reasonably extrapolated for larger system designs. The CPL heat pipe has also proved to be an efficient heat transfer device with measured evaporator and condenser film coefficients twice to four times that of conventional axial groove heat pipe designs. Other features that have been clearly demonstrated include the feasibility of multiple parallel circuits design and the ability to control operating temperature by controlling the temperature of the liquid inventory reservoir.

Areas requiring further tests and evaluation to reach firm conclusions include:

a. Pressure priming.
b. Head load sharing characteristics and limitations.
c. Sensitivity to gas bubbles entrapment in the liquid return line and feasibility of displacing gas bubbles into the reservoir.
d. Sensitivity to liquid return diameter in a 1-g operation.

Additional tests currently planned for the near future should resolve most of these questionable areas. On the basis of tests and evaluations conducted to date, the CPL heat pipe has in most aspects unquestionably proven its ability to perform as predicted. Even in questionable areas of performance, a sufficiently large degree of success has been achieved indicating a good to excellent potential for favorable resolutions of these areas. The potential levels of performance achievable within

Table 5-1. CPL Heat Pipe Development Status - February 1982

Item	Comments
1. Concept Verification	Feasibility and basic operating principles of the CPL heat pipe have been demonstrated.
2. Multiple Parallel Circuits Design	Feasibility of multiple evaporators and/or condensers in parallel has been demonstrated.
3. Heat Transport	Heat Transport capacities consistant with theoretical predictions have been demonstrated. Extrapolated Zero-g performance of the prototype CPL heat pipe with Freon-11 is 400 Watts yielding a 4000 W-M (160,000 W-in.) capacity. For ammonia, the extrapolated performance is 3400 Watts yielding a 34,000 W-M (1.36×10^6 in.) capacity. Multi-kilowatt capacity can reasonably be expected with Freon-11 in augmented system designs and an order of magnitude greater capacity with ammonia is possible.
4. Heat Transfer	Evaporator film coefficients in the range of 250-500 BTU/HR-FT^2-^{o}F have been demonstrated with Freon-11. Condenser film coefficients appear to be significantly higher. Five times higher film coefficients should be expected with Ammonia. Test set-up designed to more accurately establish evaporator and condenser film coefficients is required to verify heat transfer characteristics.
5. One-g Operation	The CPL heat pipe is sensitive to vapor header liquid entrapment in One-g. Preferred orientation and/or vapor header guard heater can illuminate this problem. Entrapment of non-condensible gas bubbles in the liquid return due to One-g buoyancy forces is also possible. Further evaluation is required to establish extend of problem and means of circumventing gas bubble entrapment.

Table 5-1. CPL Heat Pipe Development Status - February 1982 (cont.)

Item	Comments
6. Static-Wicking Height	Static wicking height capabilities have not been established.
7. Pressure Priming	The ability to prime the CPL heat pipe under load and against gravity has been demonstrated. Additional evaluation and tests are required to establish conditions which will yield repeatable and consistant priming.
8. Temperature Control	Near absolute temperature control with respect to wide sink variations has been demonstrated. Temperature control characteristics under load variations and reservior set-point temperature have also been verified.
9. Heat Load Sharing	The feasibility of heat load sharing has been demonstrated. Preferential shut-down of the condenser in the existing prototype design has been noted. Liquid subcooling into the active evaporators appear to be a limiting factor in the amount of heat that can be shared. Additional tests and evaluations are required to more clearly define subcooling limitations and to ascertain methods of minimizing or eliminating this effect.
10. Subcooling	The need for subcooling has been clearly established. Additional tests and evaluations are required to identify minimum subcooling requirements and to define design improvements to minimize subcooling requirements.
11. Non-Condensible Gas Trap	Effectiveness of non-condensible gas trap has not been demonstrated.

current state-of-the-art make it possible to consider capillary pumped two-phase heat transfer designs for large space structure applications with capacities in the ten's of kilowatt range. Furthermore, improvements and additional development in the wick material properties beyond that of currently available commercial products, may make it possible to achieve another order of magnitude improvement in kilowatt capacity, thus making it possible to consider designs with multi-hundred kilowatt capacities.

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